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No. 869

THE CONVERSION OF ENERGY IN A RADIATOR

By A. Weise

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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INTRODUCTION

The importance of the radiator has grown with the technical development of the airplane: as the engine power increases the amount of heat that must be removed grows in importance. The aerodynamic refinements on the airplane itself have made the share of the radiator on the power loss stand out more and more. The advances in high-speed flight had the same effect, since the power quota for the same installation needed to pull the radiator through the air increases as the square of the speed.

It was therefore a worthwhile task for the research to find ways to lower the cooling drag. And the success of these efforts has been so great that without this research the modern high-speed airplane would have decidedly inferior performances. These results were the outcome of painstaking exploration into the physical processes accompanying the cooling in airplane flight.

Stratosphere flying has now given the cooling problem a new motive. On the one hand, the amount of heat that must be dissipated is greater, and this applies not only to the cooling medium of the engine but also to the supercharging air and in given cases to the exhaust gas used in the turbine. Moreover, the decreasing air density renders the transfer of heat difficult, especially in the stratosphere where the outer air temperature no longer drops, that is, can no longer act against the density effect. The result is that the radiator air is heated very considerably in relation to its absolute inlet temperature. This has far-reaching results for the behavior of the radiator, which previously had been neglected on ground radiators.

*"Energieumsetzung im Kühler." Gesammelte Vorträge der Hauptversammlung 1937 der Lilienthal-Gesellschaft für Luftfahrtforschung, vorgetragen in München vom 12-14, October 1937, pp. 269-276.

The treatment of modern cooling problems calls for the coordination of thermodynamics and aerodynamics and, for sonic velocities, the kinetic theory of gases.

It is the purpose of this report to give a comprehensive discussion of all the fundamental principles and physical phenomena that offer a key to the understanding and the solution of modern cooling problems.

Behavior of Radiator Core

The design of the modern radiator lends itself to a division of the problems into behavior of the actual heat exchanger, the "radiator core," and its combined action with the airplane, as defined by the "installation."

The radiator core is an exchanger of heat designed with special regard to minimum weight. The exhibits a, b, and d of figure 1 show that the basic element of the radiator cores on the side of the cooling air is the straight tube of, in general, constant section. A number of such tubes are combined to form a honeycomb-like system. The heat taken from the cooling medium is passed on to the walls of these tubes, partly direct by scavenging, partly indirect through heat conduction, that is, fin effect. The physical conditions on the side of the cooling medium are of subordinate importance. Even the finned tube radiator of figure 1(c) may be freely considered as radiator with straight tubes. Thus, the treatment of heat exchange in the radiator is reduced to the conventional problem of heat exchange in a tube.

The radiator core mounted in an airplane acts in its installation no differently than when mounted in a closed channel on the torque stand, where it is studied for its cooling capacity and pressure losses.

Consider then the radiator block under these conditions (fig. 2). Assume for the present that the radiator is charged with cold air rather than a hot liquid, so that the velocity of the air before and behind the radiator is the same. The energy loss of the passing air due to friction manifests itself as pressure drop that can be measured with a differential pressure manometer fitted before and aft of the radiator. Now, if the radiator is used as heat exchanger, the heating of the cooling air increases the volume and, consequently, the velocity of the air along its path through the cooling passages. One effect of the increasing velocity is the increasing pressure drop due to friction. A further pressure drop is caused by the change in momentum associated with the acceleration of the air, that is, by a mass effect.

The gas-dynamic pursuance of the friction process with heat transfer in a tube indicates certain limits within which the conditions in the tube must move and which are governed by the initial condition and the initial velocity. These limits are most conveniently presented in the pressure-speed diagram, since both are of direct interest in the further treatment.

Assume that point A in figure 3 corresponds to the condition of the cooling air at entry into a heat-exchanging tube. It (A) is determined by the corresponding adiabatic curve and the pressure, or the speed. At friction without heat exchange the further conditions in the tube follow the well-known Fanno curve, whose physically potential conditions terminate on reaching sonic velocity (point B), because beyond it the entropy would have to decrease again. With heat input the conditions follow the Fanno curve. The upper limit is formed by the tangent to the adiabatic curve in the starting point of the phase. This straight line is followed if the heat is transferred without friction. It, too, has its terminal point G on reaching sonic velocity. This end point and that of the Fanno curve as well as of all intermediate phase curves have as geometrical locus a straight line through the origin of the coordinate system. The result is a closed area A B C, within which all phases of the air flow during the process of heat transfer in a tube must take place. The static temperature of the air can rise or drop during the heat transfer. On the upper straight boundary, it reaches a maximum at the pressure equal to half the pressure cut off on the ordinate axis by this straight line. The total temperature, that is, the sum of static and dynamic temperature, naturally continues to increase with heat input and reaches a maximum at sonic velocity. This presentation of the phases of the air will be employed later on.

First, we will introduce the fundamental physical connection between pressure drop and heat transfer on a wall, already recognized by Reynolds, and which affords considerable simplifications for the subsequent analysis.

Consider, to begin with, a turbulent flow, that is a flow characterized by the fact that gas bubbles are thrown back and forth transverse to the principal flow direction between the gas layers of different speed. Now assume that such a gas bubble has lost its speed on the hot wall as a result of frictional shear stresses and has through conduction taken on the temperature of the wall. At a distance from the wall the gas (air, for example) has a certain speed. Then, if the gas bubble flies through a turbulent transverse motion into this layer the following effects take place. First, there is an impact of the gas masses of different forward speed on collision, followed by loss of mechanical energy, that is, change into heat energy. This energy conversion manifests itself as a pressure loss. But another side of the process is the mixing of the gas mass heated at the wall with the cold mass of the nucleus, that is, the transport of the heat quantity absorbed by the gas bubble at the wall into the flow. It will be seen that heat transfer and pressure drop are simply different manifestations of one and the same physical process, hence that there must be some law governing the two processes.

This argument holds for turbulent as for laminar flow processes as presented for instance, in the boundary layer. The connection with the above argument is found if we analyze the individual gas molecules rather than the gas bubbles. According to the kinetic theory of gases the gas temperature is given through the kinetic energy of the irregular molecular motion. The pressure is the shock effect of this motion on the walls, which depends upon the energy as well as on the number of impinging molecules, that is, the density. Aside from the irregular motion, the molecules are superposed by the regulated motion of the flowing gas, which is likewise converted on the wall by the decelerating shock effect of the wall molecules into irregular motion, that is, arrested. Molecules thus slowed up at the wall then speed through the molecular erratic whirling velocity into the layers farther away from the wall on which the advancing motion is superposed. Thither they carry along their irregular heat motion which was increased on the wall as a result of the contact with the hotter, that is, faster-moving wall molecules due to impact. They transport heat. But they also lower the whirling energy of the molecules remote from the wall if entrained by them through shock into the general forward motion. The result is a pressure drop. Thus we find the same relationship as for the turbulent motion.

The mathematical form of the presumed law is most expediently found by dimensional analysis of the effectual physical quantities. They are: the transferred heat quantity Q (kcal/kg), the pressure drop ΔP (kg/m^2), the specific heat at constant pressure c_p (kcal/kg°), the temperature difference ΔT between wall and gas nucleus, the gas density ρ (kg s^2/m^4) and the velocity w (m/s). According to the dimensional analysis, the following expression must be constant:

$$\frac{\Delta P \ c_p \ \Delta T}{Q \ (\rho \ w) \ w} = \Omega = \text{constant}$$

$(\rho \ w)$ being the invariable stream density in the tube of constant section.

The constant Ω can, after closer analysis of the differential equations of the speed and temperature, be further defined. It is approximately equal to the Prandtl number Pr for gas, (more exactly $\Omega = 0.65 + 0.35$ Pr), that is, for air, approximately = 1. The formula shows that the heat is transferred so much more beneficially, i.e., with so much less expenditure of pressure drop, that is, on energy, as the flow velocity is lower and the temperature difference greater.

This holds, for the present, only for the heat transmission through friction. On the practical heat exchangers a divergence from this relationship is to be expected because of the concurrent fin effect and flow separation. But a mathematical check on the numerous radiator blocks measured in the closed channel indicates that the quoted expression is a constant even for practically employed blocks independent of rate of flow, depth of block and mean temperature difference (fig. 4)., which, of course, continues to diverge upwardly from the theoretical constant as the share of the pure wall friction on the heat transfer becomes less. The smooth-tube radiator with directly scavenged cooling surface manifests an almost perfect accord with the theoretical requirement, i.e., the constant is approximately equal to 1. For tubular radiators with indirect cooling surface it lies higher (at around 1.5), since part of the heat at lower temperature difference is transferred as a result of the heat conduction in the fins. The water-tube radiators with large, indirect cooling surface and disclosing flow separation at the tubes have a constant almost twice as high as the theoretical value. On these, in consequence, twice the energy must be expended for the same amount of heat transfer. In return for it, they have the advantage of less weight and less contraction of air section, so that for equal speed before the frontal surface the heat is still transferred at lower speed, that is - as the formula indicated - more propitiously.

For the ensuing investigations it therefore necessitates for each radiator block system only one single factor which completely defines its behavior with regard to the aerodynamic-thermodynamic energy conversion.

The equation deduced from the dimensional analysis for a long tube can be integrated and thus affords the law between the friction-pressure drop ΔP_r, the pressure drop due to momentum change ΔP_b, the logarithmic mean of the absolute air temperature in the tube T_m, the logarithmic mean temperature difference Δ_m between wall and air, and lastly, the block constant Ω:

$$\frac{\Delta P_r}{\Delta P_b} = \Omega \frac{T_m}{\Delta_m}$$

This relation also holds for intensive air heating, but not if approaching sonic velocity. It affords everything needed for the further radiator theory. The complete dynamical gas theorem, which retains its validity even in the range of velocity of sound, affords a differential equation which is practical for following the phases in the tube for given wall temperature and also facilitates its inclusion into the pressure-speed diagram.

This brings us to the combined effect of radiator and airplane.

Installed Radiator

The radiator installation has the purpose of rendering the rate of flow through the radiator block, which, on the installed radiator, appears as effect of the flying speed, controllable, and of keeping the external flow resistances small, especially preventing separation of air flow at the exposed surfaces of the radiator block.

To achieve the last aim, the radiator must be enveloped by walls that adapt themselves to the streamlines conditioned by the radiator and the other parts of the airplane. Figure 5 illustrates such an installation ventrally on a fuselage or engine nacelle. The cowling is so designed that the streamlines of the cooling air on entry to and exit from the cowling are parallel, thus assuring at both points the pressure of undisturbed flow. This need not be that way, as we will show later on.

The principal parts of the radiator installation are the diffuser and the nozzle. The first creates a divergent flow before the block, that is, raises the pressure conformally to Bernoulli's equation. This increase in pressure serves to overcome the pressure drop in the radiator block that - as is seen - in its installation finds itself under the same conditions as earlier in the closed channel. If the block has not absorbed the entire positive pressure produced in the diffuser a nozzle may, through convergence of the streamlines, reduce the air leaving the block to outside pressure.

One important factor for the functioning of the radiator is the rate of discharge, that is, the ratio of velocity before the radiator frontal surface w_k to the velocity of the undisturbed flow w_∞. It defines the positive pressure created in the diffuser. An energy loss within the channel is reflected in lower discharge than inflow velocity, that is, according to the equation of continuity, larger discharge section than entrance section.

In the determination of the radiator resistance, the external resistances due to skin friction on the outside of the cowling, due to separation of flow and interference with other parts of the airplane may be disregarded, since they fall under the general aerodynamics of the airplane. The remainder discussed here is termed "internal cooling resistance."

However, the transition from the consideration of energy loss in the block to the internal resistance of the installed radiator requires more than simply equating the internal energy loss defined by the pressure drop of the block, to the expendable energy which is defined through the radiator resistance and the flying speed. There are other energy quantities; for instance there is a hidden energy loss in the energy of the wake flow of the radiator, the speed of which equals the difference between flying speed and rate of discharge from the radiator. There also is a thermodynamic gain due to the heat input at higher pressure, in whose calculation the concurrently affected discharge energy must be allowed for.

It is best to avoid the presentation of all separate energies and to compute the internal radiator resistance by applying the momentum theorem, which is especially simple if it may be assumed that the pressure in the discharge section is equal to the undisturbed initial pressure. Then the integration of the momenta over the infinitely large flow section before and behind the radiator gives as resistance the product of mass in time unit passing through the radiator and the wake velocity, that is, the difference of undisturbed velocity and discharge velocity.

This method of determination affords a clear presentation in the pressure-speed diagram that retains its validity even in the case of gas-dynamics, that is, even at high velocities, because it not only affords the phases in the tube in comprehensive form but also the processes of compression in the diffuser and of the expansion in the nozzle which either appear as adiabatic curves or - when allowing for flow losses - as polytropic curves. The examples in figure 6 do not show these losses, since nozzle and diffuser can be so designed that the deviations from the adiabatic curve are small.

The starting point 1 of the compression on the adiabatic curve for the pertinent air condition lies at flying speed w_∞ (fig. 6). Following this adiabatic curve to the air inflow velocity into the tube w_k immediately gives the pressure increase in the tube. The discharge coefficient $\frac{w_k}{w_\infty}$ is plainly noticeable. The terminal point of compression 2 is simultaneously the starting point of the phase changes in the tube as previously discussed. This process must take place within the previously defined boundaries. If the radiator is without nozzle the pressure drop in the tube continues up to the discharge pressure from the radiator (point 3), which in the simplest case (as in fig. 6) is equal to the initial pressure p_∞. The case without nozzle is presented in figure 6(a), (c). If the radiator block is fitted with a nozzle, the block does not consume all of the pressure produced in the radiator. The pressure drop in the block lasts only as far as point 2', and the residual pressure gradient serves to accelerate the air in the nozzle. This speeding up process again follows an adiabatic curve which, once, as a result of the frictional pressure drop lies in the block and, then, owing to the air heating in the meantime, follows a course different from compression adiabatic curve.

Figures 6(a) and 6(b) present the case of the cold radiator. The compression from 1 to 2 is followed by the process in the block on the Fanno curve. Regardless of whether a nozzle is added (fig. 6(b)) or not (fig. 6(a)), the rate of flow through the radiator is lower than the flying speed. That is the radiator has resistance. The length 1-3 in the p-v diagram is a direct criterion for it. On the cold radiator with nozzle the resistance for equal coefficient of flow is smaller.

The compression in the diffuser does not change on the heated radiator if the coefficient of flow remains the same, but owing to the heat input in the tube, the phase curve there is to the right of the Fanno curve, which at first is noticeable on the radiator without nozzle (fig. 6(c)) in a greater discharge velocity, that is, lower resistance. If the previously cited boundaries for the phase changes are taken into account, it is found that the warm radiator, even if the friction is allowed for, may even have a negative resistance, that is, forward force. On the radiator with nozzle a flatter curve of the nozzle adiabatic in the range of higher temperatures can create a propulsive force even if the radiator block of itself because of excessive friction would have none (fig. 6(d)).

But this theoretical proof of potential radiator propulsion affords as yet no clear picture of the forces causing the propulsive force. It cannot originate in the radiator block itself because its force is opposite to the speed of flight, since frictional shear stresses only can attack its sides. Consequently, the radiator propulsion must be the resultant effect of forces on the sloping surfaces of the radiator cowling and, since - as indicated above - even a radiator without nozzle can have propulsive force, the diffuser must at least share in this phenomenon.

In the discussion of the mechanism of the origin of the propulsive force, we first consider a radiator installation without block, that is, without friction. The air in the diffuser is retarded and, as the initial pressure is to prevail again at the nozzle outlet, accelerated by the same amount in the nozzle. Rate of inflow and discharge are alike. The resistance is zero. In spite of that, forces come into play. The slowing up in the diffuser effects an internal propulsive force, which, however, is canceled by the identically great interference force of the nozzle. On the radiator with friction, that is, with installed block, the external pressure can be regained by the friction effect even without nozzle. But then the frictional stresses in the block produce interference forces and these exceed on the cold radiator the propulsive force of the nozzle, thus leaving a positive resistance. On the warm radiator but without nozzle the block manifests - as previously shown - aside from the friction, a further cause of pressure drop, namely, the mass effect of the air on accelerating following the volume increase as a result of heating. This pressure drop makes it possible to regain the outside pressure again without necessitating corresponding wall forces in the block or in a nozzle, since the support occurs on the accelerated mass. The corresponding part of the diffuser propulsive force is not compensated, that is, remains effective. If the friction effects are small enough it may lead to a positive propulsion. The warm radiator with nozzle receives yet an additional propulsive effect which fundamentally is created in the same manner.

The radiator cowling in figure 5 presents merely a special case, namely, of parallel flow in the inlet and outlet section. But it is not at all necessary and it practically never happens that the flow there is parallel, that is, the pressure equal to that of the undisturbed flow.

The next discussion is on the effect of this deviation on the energy conversion and the force effect on the radiator.

Figure 7 compares several cases, that of figure 5 being shown as figure 7(a). The shaded area in the radiator outlet indicates the mixing zone where the air layers of the outside air moving at different speeds have exchanged momentum with the air already passed through the radiator. By this exchange of momentum, the mechanical energy is transformed to heat energy. In spite of that, there is no additional resistance in this case because the total momentum remains constant during the mixing process.

Figure 7(b) shows the same radiator with shortened diffuser, that is, enlarged inlet section. The question is whether in this case the rate of flow changes and how the streamlines proceed. Since the conditions of flow at the radiator exit are as before determined by the form of the nozzle cover and the conditions within nozzle and block themselves are the same, the radiator requires as before the same velocity and the same pressure before the frontal area of the block. Now, is the shortened diffuser still in position to meet this requirement? It is, because the air before the inlet area in the free flow is dammed up as a result of which the streamlines themselves diverge as if the original diffuser length still existed. This streamline flow is made possible through the effect of the diffuser wall which then acts similarly to an airfoil. If the flow is the same as that on the unshortened diffuser, then the force effects must also be the same. Since only a part of the forces can appear on the inside of the diffuser, because of elimination of part of the wall area, the remaining part must be produced at the outside. This comes to pass through the flow around the front of the cover at increased velocity, that is, lower pressure. This negative pressure produces the part of the propulsive force of the diffuser formerly created on the inside. This increase of velocity also naturally increases the hazard of separation of flow on the outside, and must be counteracted by suitable design. According to test data the shortened diffuser in conjunction with high compression has a substantially better efficiency than the unshortened diffuser.

Figure 7(c) shows the shortened diffuser and shortened nozzle. The discharge here takes place at a pressure in excess of the undisturbed outside pressure. Here also the boundary streamline could follow the same course as if the wall existed, provided that, in constrast to the conditions on the diffuser as a result of the velocity difference between the air leaving the radiator and that flowing past at the outside, no mixing would be provoked again. It should be noted that the mixing in this case influences the inside cooling resistance, whose magnitude and sign depend upon whether the mixing occurs at positive or negative pressure, on the magnitude of the masses sharing on the mixing, and on the extent of the heating of the air in the radiator.

In general the undisturbed outside pressure is reached again before any appreciable mixing of air masses takes place, so that the mixing effect on the internal resistance amounts to very little. The considerations regarding the shortened diffuser also afford the key to the understanding of the control of a radiator. It was shown that the size of the entrance section had no effect on the flow since it is determined by the nozzle. It is therefore possible also to regulate the flow of the radiator from the nozzle without diffuser adjustment.

Figure 8 shows a radiator with three different control settings. The diffuser is fixed while the nozzle in all three settings on the outside is so designed that it does not affect the flow on the exit due to the other parts of the airplane; the air can always leave the radiator at the same pressure. Thus the ratio of exit section of the nozzle and of the inlet section of an assumedly whole diffuser is constant at all control settings. But, as the diffuser inlet section does not change the course of the streamlines before the diffuser must conform to the cited conditions. Enlarging the nozzle to increase the flow (b) reduces the divergence of the streamlines before the diffuser, that is, the pressure rise diminishes. Ultimately the point is reached where the diffuser acts as a whole diffuser, that is, does no longer dam up in the free stream. On opening the nozzle still further (c), the streamlines converge before the diffuser, which is now too long, or in other words, has a too narrow entrance section, and negative pressure prevails in the inlet. Then the diffuser must overcome within a pressure rise greater than the effective pressure. This, of course, entails unnecessarily large losses which lower the flow relative to the diffuser with little loss and increase the resistance. This behavior of the streamlines must be forestalled.

General Radiator Theory

From the foregoing arguments a general radiator theory can be developed that can be used in conjunction with the pressure-velocity diagram in the treatment of a radiator on the airplane, even with gas-dynamical generalization, and in answering all quantitative questions theoretically for any heating, provided the velocities are not abnormally high. On experimental bases only the cited constant of the radiator block system containing the relation between heat transfer and pressure drop is required. With this constant, the pressure losses at any air heating can be directly assessed without having to resort to individual test data. One thereby eliminates the tedious repeated computations and first attains in fact the analytical solution of the problem. Having then computed the internal radiator resistance according to the previously described methods, the actual test is reduced to determining the radiator block constants and the external radiator resistances by the customary aerodynamic methods. The depth of the block itself must be so proportioned that the computed air heating or the computed pressure drop occurs. This theory supplies the answers to radiator problems so far as it relates to a combination of aerodynamics, thermodynamics, and gas dynamics. The results of such a theory are illustrated on a few examples.

Figure 9 gives for a radiator block constant $\Omega = 1$ the curve of the relative air heating in the radiator block η (ratio of temperature rise to initial temperature difference between air and wall) plotted against the coefficient of flow η_{ae} (here the ratio of velocity in the cooling channel to flying speed for a diffuser efficiency = 1) for different nondimensional initial temperature differences ξ (ratio of absolute cooling surface temperature to absolute air temperature at entry into the radiator block). $\xi = 1$ for the cold radiator, about $\xi = 1.2$ for the water radiator at sea level, $\xi = 1.8$ for the high-temperature liquid radiator in the stratosphere, and around $\xi = 5.5$ for an exhaust cooler in the stratosphere.

Figure 10 presents for $\Omega = 1$ the curve of the nondimensional resistance Ω_e of a radiator without nozzle plotted against the coefficient of flow. The nondimensional resistance is no resistance coefficient of the frontal radiator surface, but is at constant ξ directly proportional to the resistance* that the airplane must overcome

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$$^{*}\; W = \Omega_e \frac{Q\, w_\infty}{g\, c_p\, \Delta T}$$

where:

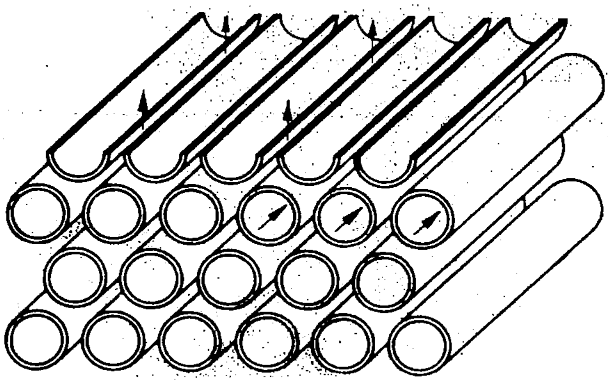
W is resistance (kg)
Q, heat dissipation (kcal/s)
w_∞, flying speed (m/s)
g, gravity
c_p, specific heat of air
ΔT, initial temperature difference between wall and air

to carry off a certain amount of heat. It therefore affords a better criterion than the drag coefficient, which necessitates first the determination of the frontal area of the radiator before the actual resistance is obtained. The curves of the example in figure 10 are indicative of a certain coefficient of flow at which the resistance becomes minimum. This fact, confirmed here in theory, has been previously established in experiments. The explanation for the minimum is that, at very high rates of flow, the resistance becomes greater because the heat is carried off at a high rate, that is, uneconomically, while at low rates the greater heating of the cooling air consistently reduces the mean temperature difference, and so lowers the economy of heat removal.

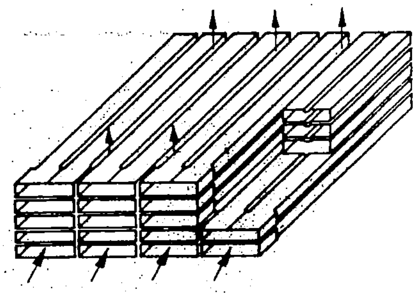
Another general result of the radiator theory is as follows: the power loss quota referred to engine output increases at equal coefficient of flow and equal nozzle setting at the square of the flying speed while the flying height exerts an influence on the cooling resistance through the absolute outside temperature but not through the pressure or density. On the other hand, the variable air density affects the size and consequently the weight of the radiator.

These outlines of a general radiator theory should be sufficient to indicate what steps the subsequent elaboration of the fundamental concepts must follow to assure its practicability.

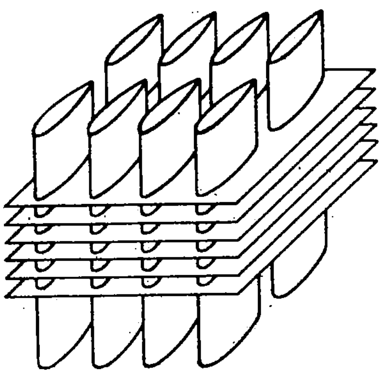
Translation by J. Vanier,
National Advisory Committee
for Aeronautics.



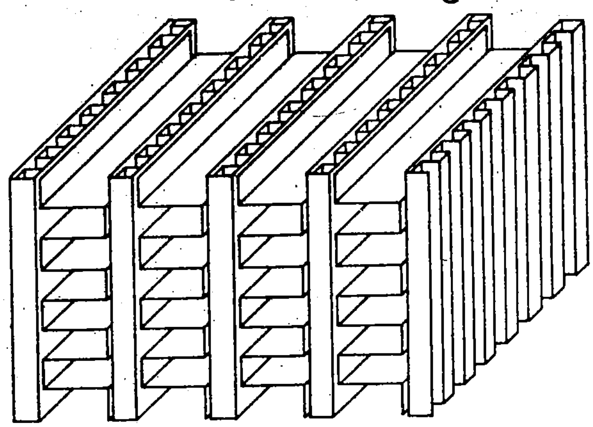
(a) Air-tube radiator without indirect cooling surface.



(b) Flat-tube radiator with indirect cooling surface.



(c) Finned-tube radiator.



(d) DVL-air radiator.

Figure 1.- Types of block radiators. (the cooling air is horizontal from front to back).

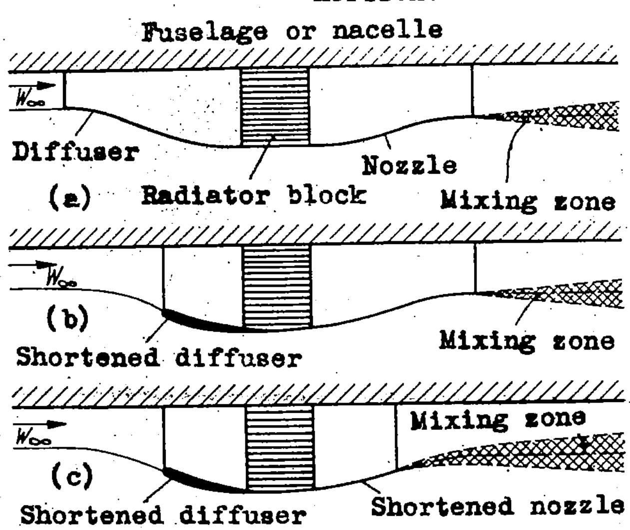


Figure 7.- Radiator installation. (change of flow following modified inlet and outlet section).

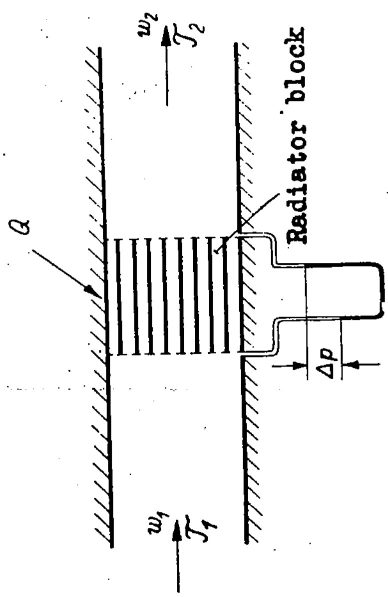
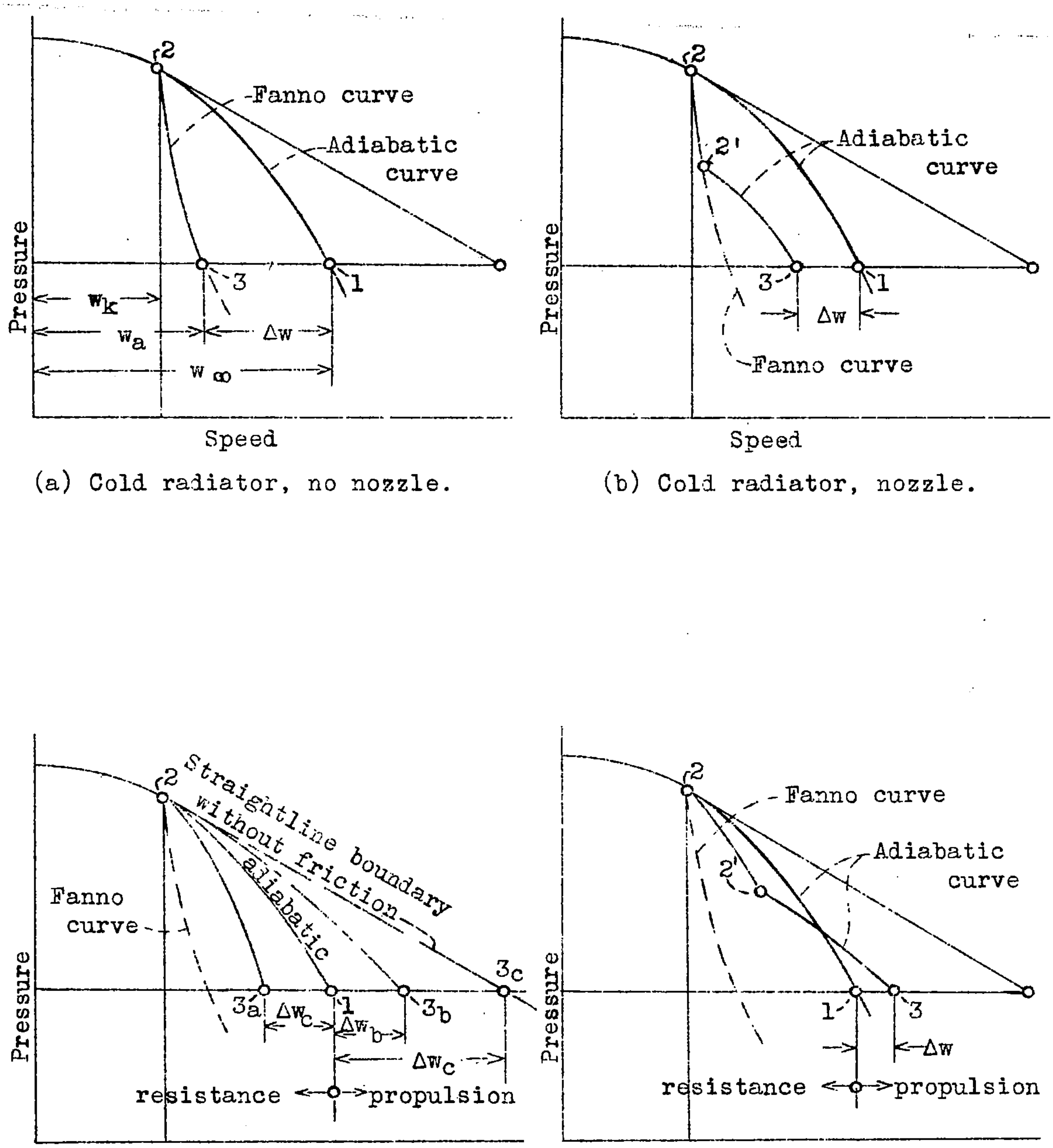


Figure 2.- Radiator block in closed channel.



(a) Cold radiator, no nozzle. (b) Cold radiator, nozzle.

Speed Speed

(c) Warm radiator, no nozzle. (d) Warm radiator, nozzle.

Figure 6.- Working process of radiator in the pressure-speed diagram.

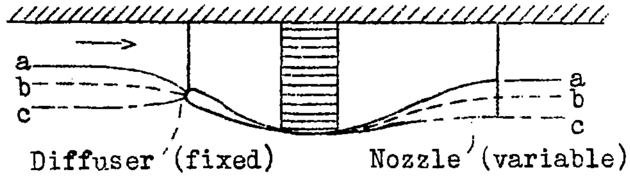


Figure 8.- Control of installed radiator thru adjustment of nozzle with fixed diffuser.

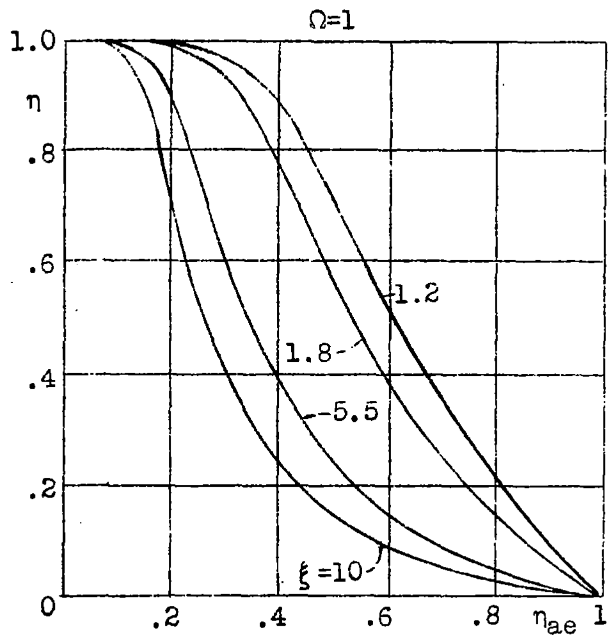


Figure 9.- Heating of air in the radiator.

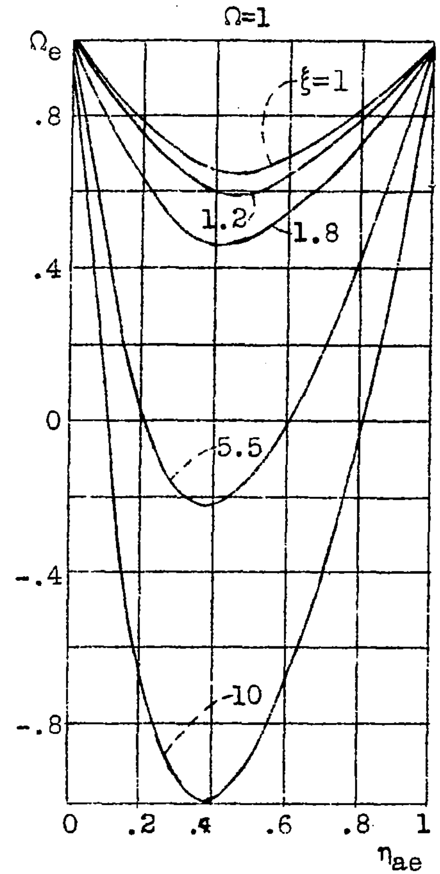


Figure 10.- Nondimensional resistance of radiators without nozzle.